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HYDRAULIC INDUCED INSTABILITY ON A VERTICAL SERVICE WATER PUMP: CASE HISTORY

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The case history contained in this paper provides insight toward the mechanical and hydraulic behavior of a vertical pump. It clearly demonstrates the need for measurements on the rotor at or near the impeller area.

INTRODUCTION

This case history reports the results of an analysis on a service water pump. This pump is typical of the water pumps used throughout the power generation industry. Although little is known of the mechanical behavior of vertical pumps because of difficulty in modeling the rotor system, recent developments in the application of submersible proximity transducers have made possible the measurement of pump dynamics under operating conditions.

The purpose of this study was to determine the proper selection and installation of vibration-monitoring transducers as well as to measure the effects of imbalance, misalignment, and hydraulics on the performance and reliability of vertical pumps. In addition, the cause of shaft failures on this pump was to be determined.

MACHINE DESCRIPTION

An outline drawing of this pump and motor is shown in figure 1. A number of shafts on this specific pump had failed in recent years for unknown reasons. The lower bearing consists of a grease-lubricated bronze bushing supported by the pump end bell. Intermediate "bearings," supported by the pump vertical column, are water lubricated and lined with scalloped rubber. This type of bearing is sometimes referred to as a "cutlass" bearing or bumper. The couplings between the shaft sections are rigid and transmit torque through two keys spaced 180° apart. For example, the transducer identified as DWD in figure 1 is located at the bottom bearing and is the west proximity probe. The rated pump performance is

(1) Speed, 720 rpm
(2) Horsepower, 1250 bhp
(3) Rated flow, 30 000 gal/min
(4) Rated differential pressure, 140 ft

This pump is one of three identical pumps connected to a common header. Flow is regulated by a control valve in the discharge piping of each pump.

TRANSDUCER SELECTION

Since very little information is available on the rotordynamic characteristics of this type of pump, proximity and velocity transducers were installed at each bearing and at the motor coupling. Locations are shown on the pump outline drawing. The transducers are identified by a three-letter code shown in the legend in figure 1. Transducers at locations C and D were below the river level and had to be waterproof. The transducer also had to be designed to provide mechanical protection and to survive the abrasive action of sand in the flowing water. Figure 2 is an outline drawing of the probe design used.

STATIC TESTING

The rotor system was tested experimentally for both lateral and torsional resonances. Testing was performed by suspending the instrument pump shaft, motor rotor, and pump impeller vertically and perturbing the assembly by striking it with an instrumented force hammer. The results of these tests, summarized in table 1 and figure 3, were correlated with data obtained in later field testing. The pump speed of 720 rpm dictated that the system would run above the second and below the third lateral balance resonance.

The first torsional resonance of the rotor occurs in the range 690 to 765 cpm. Since this range includes the running speed, torsional analysis was warranted. Strain gauges with telemetry were employed to measure shear strain. Values were extremely low - well below the fatigue limit - indicating a system well damped torsionally.

FIELD TESTING

The pump impeller was modified to allow the addition of balance weights to the impeller blades, without pulling the pump, by using a scuba diver to swim into the pump intake to add or remove weights. The pump impeller was then shop balanced to critical standards and installed in the pump case. The pump was assembled with the transducers mentioned earlier. Extreme care was taken to align the pump and motor. The axial position of the pump was set to the manufacturer's specification, establishing the clearance between the leading edge of the impeller and the casing.

IMBALANCE

The balance condition of the impeller was changed while holding the other variables constant. The amount of imbalance added, 80 oz-in, resulted in a centrifugal force equal to 84 lb. For a rotor weight of 1000 lb, this was 8.4 percent of the rotor weight.

Figures 4 and 5 show orbit and time-base plots of trial runs with and without the imbalance weight. These measurements were taken at the lower bearing (location D) and at location B. Note that the plots at the lower bearing (D) clearly show the effects of unbalance but the plots at the upper bearing (B) do not show a significant change.

MISALIGNMENT RESPONSE TESTING

The alignment between the motor shaft and the pump shaft was changed by adding a 0.010-in shim under one side of the motor. This resulted in an angular misalignment of approximately 2°. Radial offset is impossible with this type of coupling. Orbit and time-base plots of data at location B are shown in figure 6. The slight elliptical shape of the orbit on the second trial run indicates a preload caused by the misalignment. Spectrum plots of the same data (fig. 7) do not give strong indication of misalignment. Since the misalignment manifests itself as 1x vibration, no strong indication of traditional 2x misalignment is indicated on the spectrum plot. The preload was not as apparent at the bottom bearing, as expected.

HYDRAULIC RESPONSE TESTING

The pump was run with two different clearances between the leading edge of the impeller and the casing. Tests were conducted with design clearance of 0.043 in and excessive clearance of 0.200 in. The discharge pressure decreased by 20 psig with the increased impeller clearance, which is a good indication of the amount of recirculation and efficiency losses. With design clearance, spectrum plots of proximity probe data from the lower bearing showed strong subsynchronous response at 372 cpm (fig. 8). This frequency is the first lateral natural frequency (critical) of the pump shaft. When the clearance was increased, the subsynchronous vibration became less stable and occurred over a broader frequency range. As observed earlier with the balance response testing, the transducers located near the motor were unable to measure the subsynchronous activity on the impeller.

The hydraulic response testing determined that the excitation of the first balance resonance by the subsynchronous vibration was the cause of the recent shaft failure. These failures were diagnosed as bending fatigue failures by metallurgical analysis. However, several unanswered questions remained:

(1) Why was the vibration higher with design clearance?
(2) Why did the pumps operate without shaft failures until recently?
(3) Why was the discharge pressure only 62 psig when rated pressure at present river level is 85 psig?

When maintenance records were researched, it was found that the start of shaft failures was preceded by a change in the supplier of the impellers. Recent impellers were not purchased from the original equipment manufacturer (OEM). The OEM has indicated that subtle changes in the blade attack angle could cause the reexcitation of the first lateral natural frequency (critical) because of vortex shedding.

CONCLUSIONS

Two major conclusions resulted from this testing:

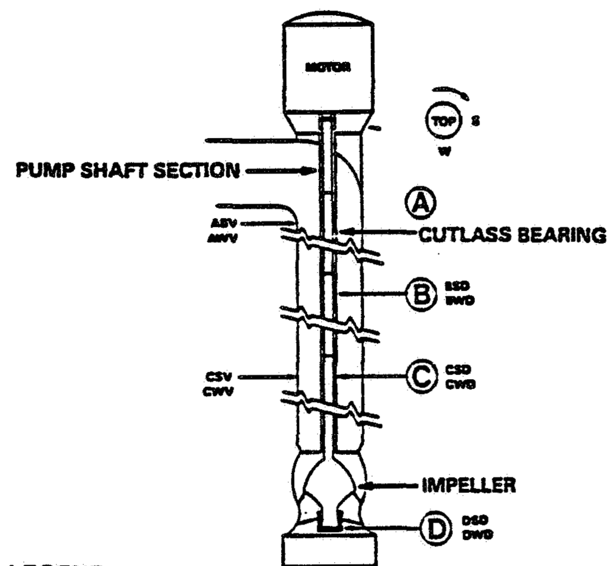
1. Long vertical pumps cannot be adequately monitored without measuring the motion of the shaft near the impeller. This pump was analyzed on many occasions by using transducers installed at location A. These casing transducers (velocity and

acceleration) failed to detect the large subsynchronous vibration occurring at the first balance resonant frequency of the shaft with the pump at running speed (720 rpm).

2. Hydraulic effects appear to be several orders of magnitude higher than imbalance and alignment effects. Any diagnostic effort should include the correlation of hydraulic data.

TABLE 1

ROTOR NATURAL FREQUENCIES (CPM)		
MODE	LATERAL	TORSIONAL
1	375	690
2	525	—
3	975	—
4	2325	—
5	3375	—
6	4275	—
7	5475	—
8	6600	—



LEGEND

First Letter: Vertical location of measurement point (i.e., A, B, C, etc.)

Second Letter: Horizontal location ("S" for south side of pump, "W" for west)

Third Letter: Type of transducer ("D" for displacement proximity probe and "V" for velocity transducer)

Figure 1. - Vertical low-pressure service water pump.

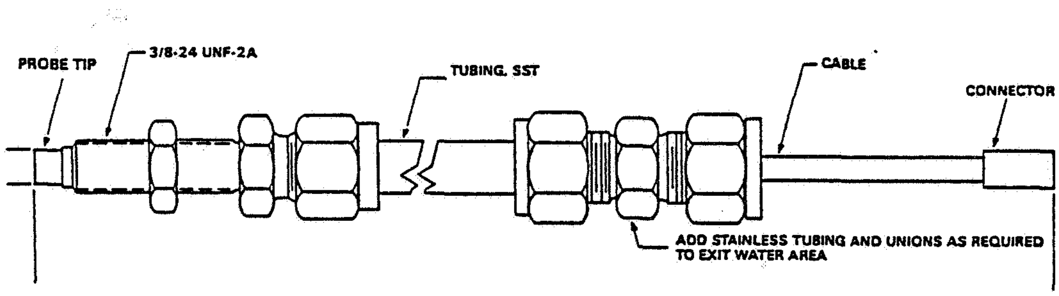


Figure 2. - Waterproof proximity probe transducer.

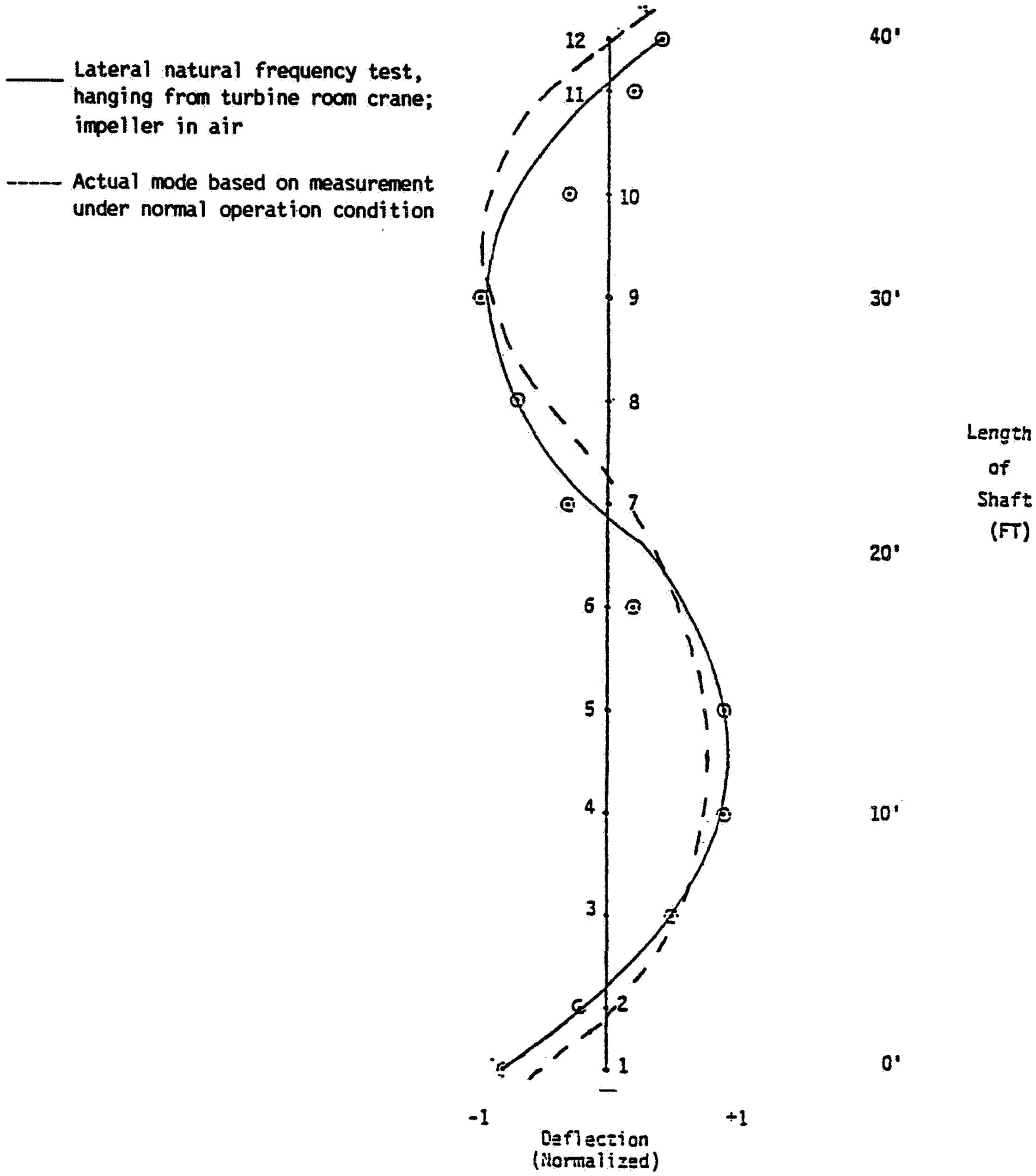


Figure 3. - Mode #2 at 525 cpm.

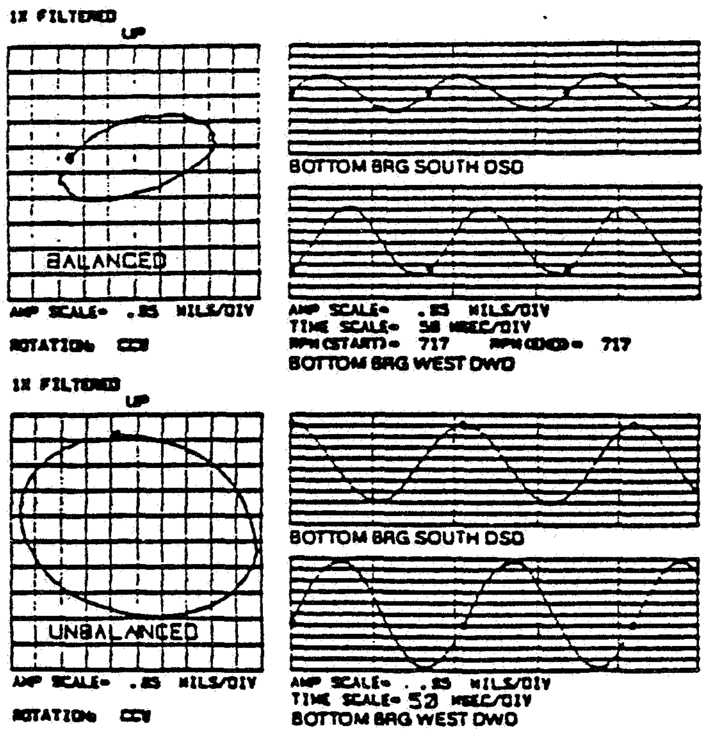


Figure 4. - Bottom-bearing orbit and time-base plots with and without imbalance weight.

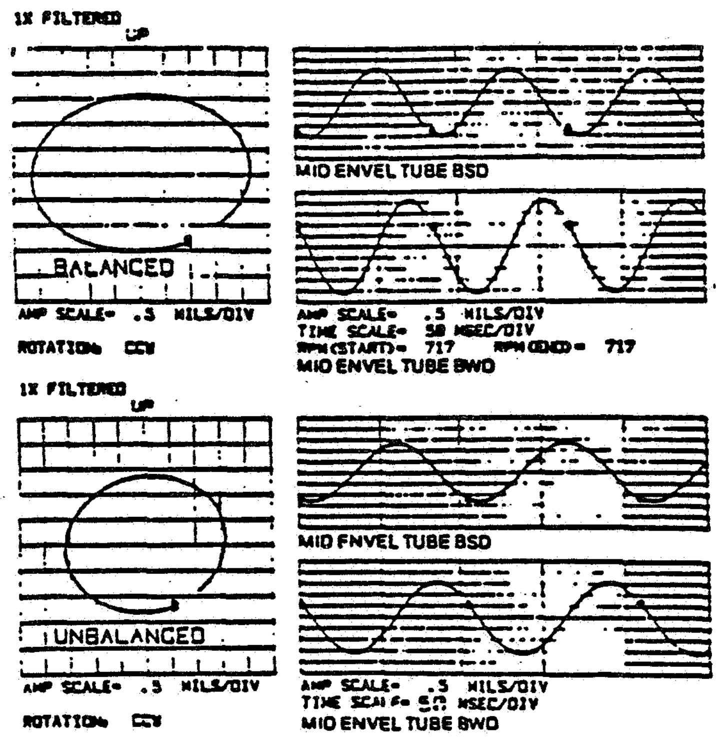


Figure 5. - Upper-bearing orbit and time-base plots with and without imbalance weight.

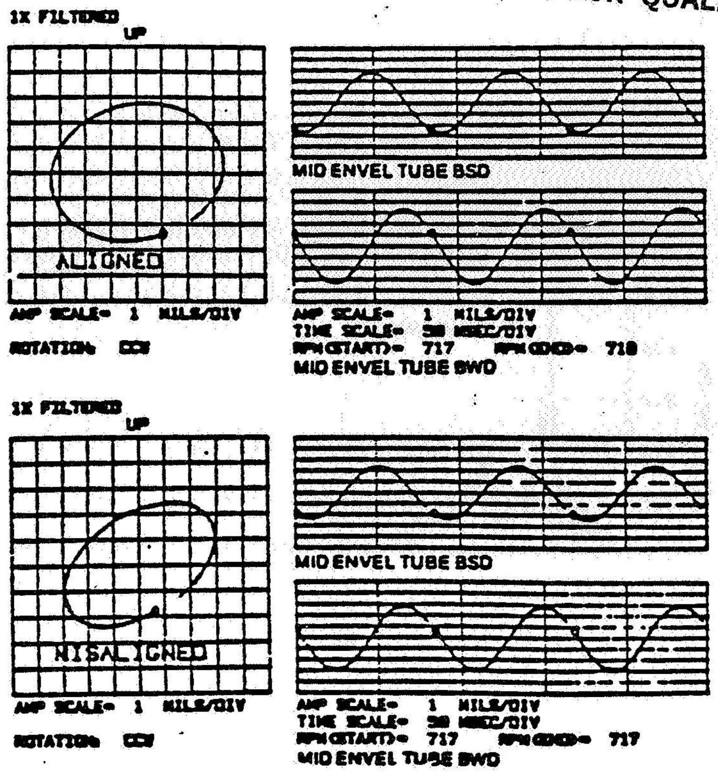


Figure 6. - Upper-bearing orbit and time-base plots with misalignment.

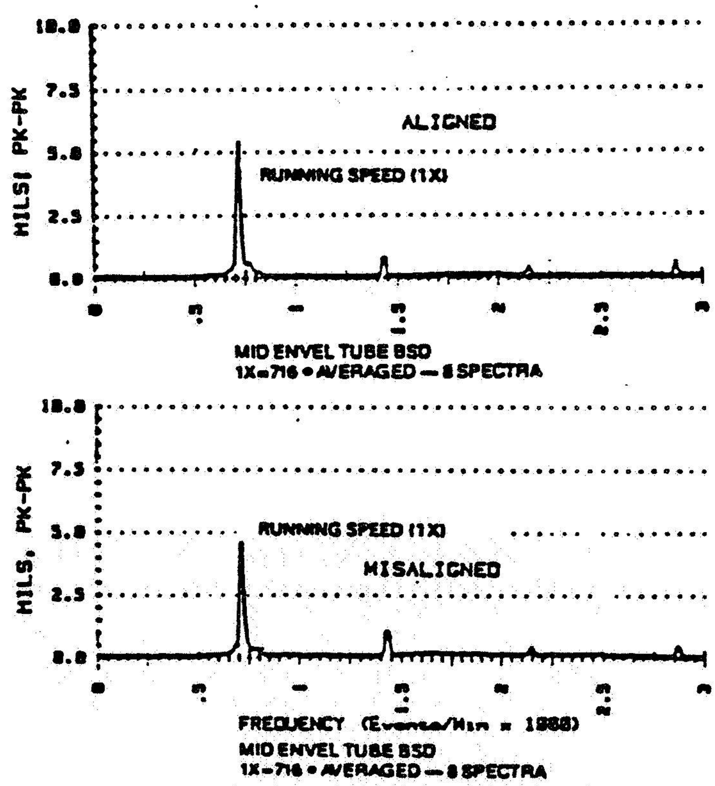


Figure 7. - Upper-bearing spectrum plots with misalignment.

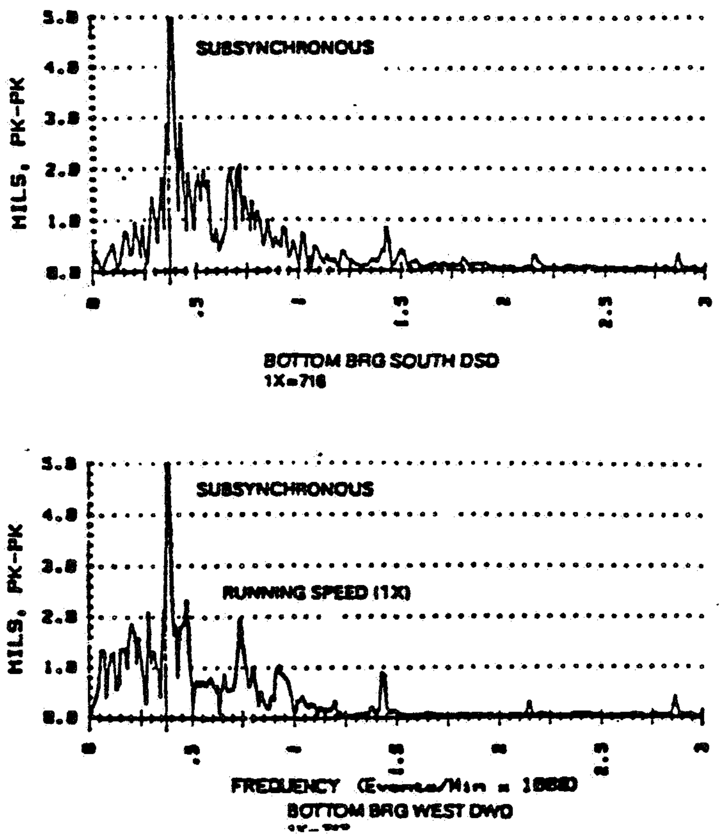


Figure 8. - Bottom-bearing spectrum plots showing subsynchronous vibration component.

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